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Kiyoshi Sawai
Hiroshima Institute of Technology, Japan, sawai@me.it-hiroshima.ac.jp

Doi Manabu
Hiroshima Institute of Technology, Japan, m131404@cc.it-hiroshima.ac.jp

Noriaki Ishii
Osaka Electro-Communication University,Japan, ishii@isc.osakac.ac.jp

Noboru Iida
Panasonic Corporation,Japan, iida.noboru@jp.panasonic.com

Kenji Kinjo
Panasonic Corporation,Japan, kinjo.kenji@jp.panasonic.com

Study on Flow Characteristics of Oil Viscosity Pump for Refrigerant Compressors

Kiyoshi SAWAI[1]*, Manabu DOI [1], Norimaki ISHII [2], Noboru IIDA [3], Kenji KINJO[3]

[1]Hiroshima Institute of Technology, Dept. of Mechanical Engineering
Hiroshima, Japan
Phone: +81-82-921-5475, E-Mail: sawai@me.it-hiroshima.ac.jp

[2]Osaka Electro Communication University, Osaka, Japan
Tel/fax: +81-72-820-4561

[3]Panasonic Corporation, Appliances Company, Corporate Engineering Division
Kusatsu, Shiga, Japan
Tel: +81-77-562-5658, fax: +81-77-563-1967

* Corresponding Author

ABSTRACT

Reciprocating compressors are commonly used in household refrigerators, and lubricants is supplied to the compression mechanism by the oil viscosity pump. While it is expected that this lubricants has a great influence on the compressor's performance and reliability, its flow characteristics is does not mean elucidated. In this study, we examined the flow characteristics of the oil viscosity pump that had a spiral groove on the shaft surface opposite to the bearing surface, using both theoretical and experimental study. The results revealed followings:

(1) When oil flow in the oil groove is assumed to be a "uniform, two-dimensional flow which is a steady and sufficiently developed laminar flow," the theoretical flow rate in the oil viscosity pump and the pump head can be expressed with a linear function formula. It was also confirmed in experiments that both were in a linear function relationship.

(2) Whether the theoretical flow rate formula can be applied does not depend on the oil groove angle, but on the oil groove length. It is possible to estimate the flow characteristics within an average relative error of 30% if the oil groove length is 60 mm or larger.

(3) Whether the theoretical flow rate formula can be applied depends on the oil groove depth if the oil groove has a rectangular cross section. It is possible to estimate the flow characteristics within an average relative error of 30% if the aspect ratio of the oil groove is 0.5 or smaller.

1. INTRODUCTION

Recently, refrigerators account for 14% of the total electricity consumed by households in Japan. In addition, the compressor accounts for 80% of the electricity consumed by a refrigerator. As shown in Figure1, suction and compression of refrigerants are conducted as the piston makes a reciprocating motion within the cylinder, and reciprocating compressors which have relatively high efficiency are commonly used in refrigerators. In the reciprocating compressors, lubricants is supplied to the compression mechanism from the bottom of the shell by the oil pump system. This oil pump system is comprised of [1] a centrifugal pump comprising the cylindrical part and a cap attached at the bottom end of the shaft, [2] a reed pump with a blade placed in the cylinder, and [3] the oil viscosity pump which supplies oil using the shearing stress generated on the bearing's surface as a spiral groove is formed on the surface of the shaft opposite to the bearing surface. The lubricants supplied by this oil pump system is expected to have a great influence on the performance and reliability of the compressor, then its flow characteristics have been studied by several researchers. Asanuma (1951) made a theoretical analysis on the flow characteristics of the oil viscosity pump, but the influence of gravity is not taken into consideration. Weifeng *et al.* (2010) made a numerical simulation of the centrifugal pump of the reciprocating compressor. Kim *et al.* (2002) and Luckmann *et al.* (2009) made a numerical analysis of centrifugal and oil viscosity pump in reciprocating compressor. Alves *et al.* (2010) made a numerical analysis of screw pump system attached to shaft-end for reciprocating compressor. However, in these studies, flow characteristics (relationship between oil flow rate and pump head) of the oil pump system has not been clear. In this study, we focused on the oil viscosity pump which is expected to have the largest influence on the flow characteristics out of all the oil pump system's components, and examined its flow characteristics using both theoretical and experimental study.

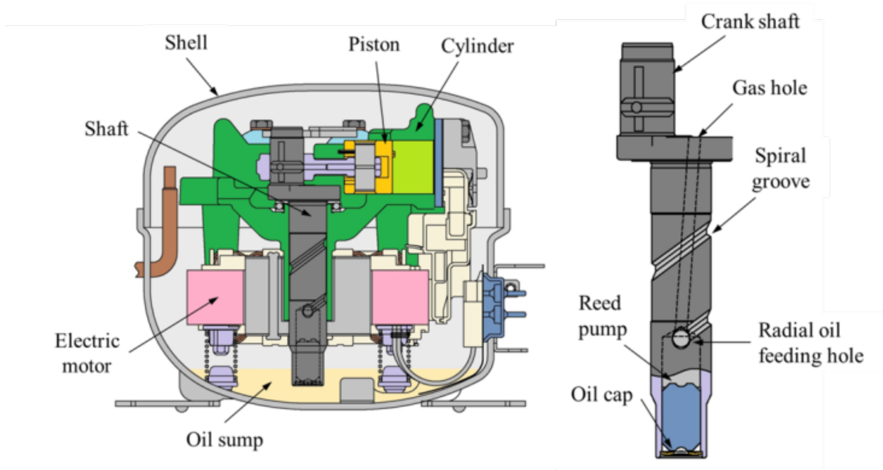

Figure 1: Compressor for household refrigerator

2. THEORETICAL STUDY OF OIL VISCOSITY PUMP

2.1 Overview of and coordinate system for oil viscosity pump

Figure 2 shows a schematic diagram of the oil viscosity pump. It shows that the shaft has a spiral groove with a rectangular cross section. A schematic view of the oil groove is also shown in Figure 2. The bearing surface moves relatively to the right direction when the shaft rotates to the left. In this case, a viscous shearing force acts on the oil inside the oil groove on the surface in contact with the inner wall of the bearing, which makes the oil climb up along the oil groove as a consequence.

The length of this oil groove which functions as a viscosity pump was designated as L, the length of the oil groove in the direction of the shaft axis L_s, and the angle of the oil groove attachment β. In addition, the oil groove was formed with a rectangular cross section with a shallow depth h and width b. And as the coordinate system for theoretical study, the direction of the oil flow in the oil groove was defined as the x axis, the direction of depth in the oil groove as the y axis, and the direction of width in the oil groove as the z axis.

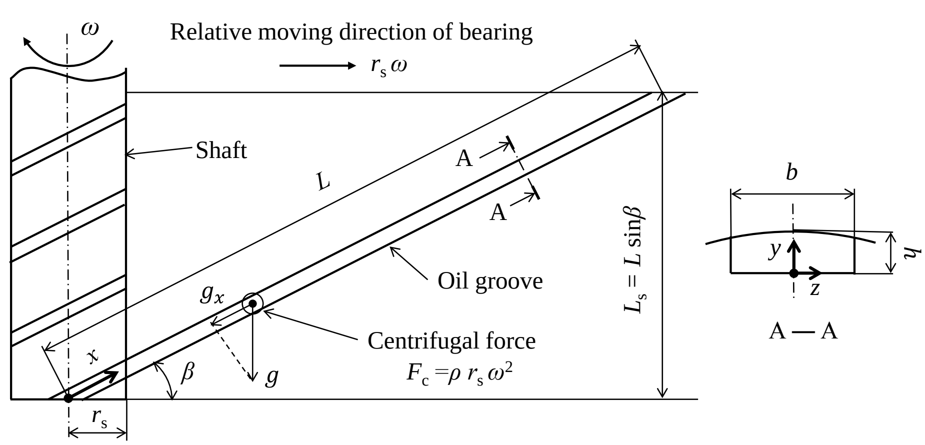

Figure 2: Schematic view of the oil viscosity pump

2.2. Fundamental equations for oil flow in spiral groove

In theoretical study, we introduced two assumptions to the oil flow inside the groove. The first assumption was that the oil flow was a steady and sufficiently developed laminar flow in the direction of the x axis. The second assumption was that the oil flow had a velocity component in the z axis direction $w = 0$ on each cross section of the oil groove, and was a uniform $x - y$ flow in z direction. In this theoretical study, it was also assumed that the leakage through the clearance could be neglected as the clearance between the shaft surface and bearing was sufficiently small compared to the oil groove depth h. The equation of continuity in Eq.(1) and Navier-Stokes formula in Eqs.(2) and (3) were applied to this $x - y$ flow in oil groove. In this case, it was taken into consideration that the centrifugal force F_c was applied to the oil inside the oil groove as a body force.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = g_x - \frac{1}{\rho}\frac{\partial p}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = g_y + \frac{F_c}{\rho} - \frac{1}{\rho}\frac{\partial p}{\partial y} + \nu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

2.3. Oil flow Velocity *u* and flow rate formula

Oil flow velocity u inside the oil groove as shown in Eq.(4) is derived by applying two assumptions in the previous section to Eqs.(1), (2) and (3) and integrating Eq.(2) twice:

$$u = \frac{y}{h}\left\{U - \frac{h^2}{2\mu}\left(\frac{\partial p}{\partial x} + \rho g \sin\beta\right)\left(1 - \frac{y}{h}\right)\right\} \tag{4}$$

Figure 3 illustrates the oil flow velocity u expressed by Eq.(4). The bottom end ($y = 0$) in Figure 3 indicates the bottom of the oil groove, and the top end ($y = h$) the bearing inner wall. The broken line corresponds to the Couette-flow, and the solid line shows velocity distribution u that is suppressed by the pressure gradient $\partial p/\partial x$ and gravity g acting on the flow compared to the Couette-flow.

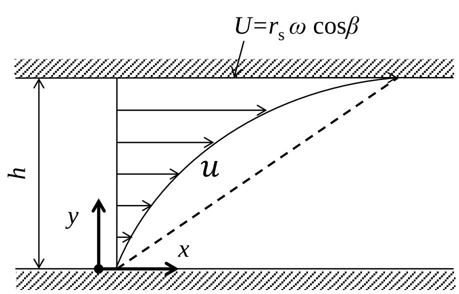

Figure 3: Velocity distribution in spiral groove

Here, pressure gradient $\partial p/\partial x$ in Eq.(4) is expressed by Eq.(5) using pump head H. Therefore, by substituting it in Eq.(4), integrating Eq.(4) until y=0 → h, and multiply the oil groove width b, Eq.(6) can be used to express the theoretical flow rate of oil Q_t flowing inside the oil groove.

$$\frac{\partial p}{\partial x} = \frac{\rho g H}{L} \tag{5}$$

$$Q_t = \frac{bh}{6}\left\{-\frac{h^2 g}{2\nu L}H + 3U - \frac{h^2}{2\nu}g\sin\beta\right\} \tag{6}$$

Based on Eq.(6), it was revealed that there was a linear functional relationship between the theoretical flow rate Q_t and pump head H.

3. OUT LINE OF THE EXPERIMENT

3.1. Experimental apparatus

We prepared some evaluation apparatus to examine the oil viscosity pump's flow characteristics as shown in Figure 4. In this experiment, lubricants with a viscosity close to the actual operation viscosity on refrigeration cycle at room temperature was selected and stored in the oil bath. Shafts equipped with a spiral groove on the surface are prepared to suit the purpose of each experiment. When this shaft is rotated, oil is pumped up along the oil groove formed on the shaft surface. By pulling out the oil into a tube from the top of the bearing and giving resistance to the flow with a valve installed in the middle of the pipe line, pump head H was applied to the oil viscosity pump in this system. During experiments on oil flow characteristics, pump head H was read on a scale set up at the head-measuring section, and the experimental oil flow rate Q_e was measured using an electronic balance.

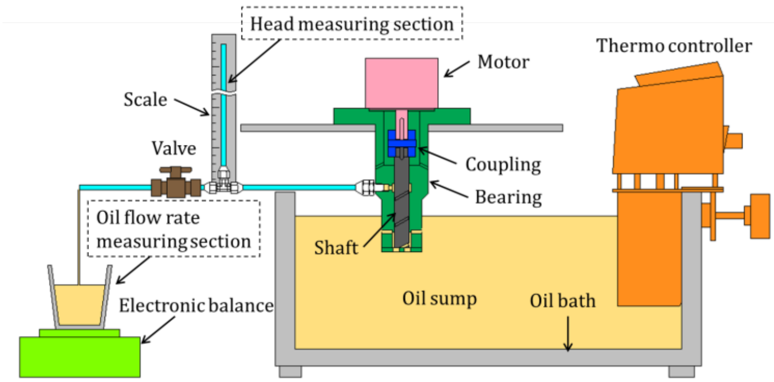

Figure 4: Experimental apparatus

3.2. Description of experiments

To find out the applicable range for the theoretical flow rate formula for the oil viscosity pump in Eq.(6), we conducted three types of experiments as shown in Tables 1 to 3. The purpose of experiments in Tables 1 and 2 was to study whether Eq. (6) was applicable or not when what value of oil groove length L was ensured, and flow characteristics is studied by varying each of the shaft length L_s and oil groove angle β. In the experiment in Table 3, we studied the relationship between the oil groove aspect ratio h/b and the possibility of application for Eq.(6) by varying the oil groove width b and depth h while keeping the oil groove's cross section constant. Furthermore, a shaft with a $\phi 18mm$ diameter was used in all experiments.

Table 1: Shaft specifications 1 (various oil groove length)

No.	b, h [mm]	β [°]	L_s [mm]	L [mm]
1	3.0, 1.0	72	100	105.2
2			75	78.9
3			50	52.6
4			30	31.5

Table 2: Shaft specifications 2 (various oil groove angle)

No.	b, h [mm]	β [°]	L_s [mm]	L [mm]
3	3.0, 1.0	72	50	52.6
5		61		57.2
6		54		61.8
7		43		73.3
8		30		100.0

Table 3: Shaft specifications 3 (various rectangular cross-section of oil groove)

No.	b [mm]	h [mm]	β [°]	L_s [mm]	L [mm]
9	3.5	0.85	54	50	61.8
6	3.0	1.0			
10	2.5	1.2			
11	2.0	1.5			
12	1.5	2.0			

4. EXPERIMENTAL RESULTS

As typical experimental results, the flow characteristics of Shaft No.8 are shown in Figure 5. The blue line in the figure indicates the experimental flow rate Q_e, and the red line the theoretical flow rate Q_t calculated using Eq.(6) for the same head. Based on this result, it was confirmed that the tendency matched both the theoretical flow rate and experimental flow rate and that there was a linear functional relationship between oil flow rate and pump head. It showed that the theoretical flow rate formula in Eq.(6) expressed flow characteristics of the oil viscosity pump that were close to the actual situation.

In this study, we also evaluated the difference between the flow characteristics obtained in the experiment and the flow characteristics calculated from the theoretical flow rate formula Eq.(6) by using the average relative error E_{ave} defined in Eq.(7).

$$E_{ave} = \left(\sum_{i=1}^{n} \frac{|\delta_{e,i} - \delta_{t,i}|}{\delta_{e,i}} \right) \Big/ n \times 100 \ \ [\%] \tag{7}$$

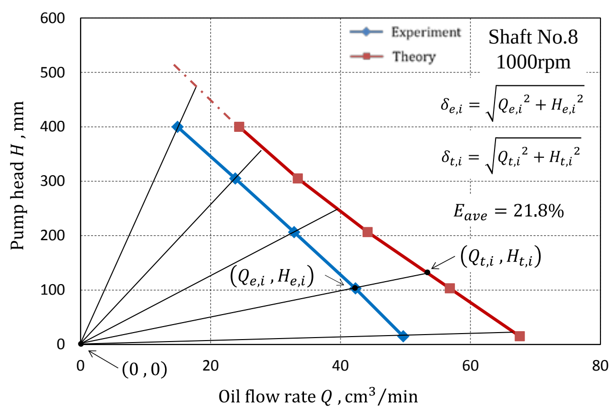

Figure 5: Experimental results (representative example)

$\delta_{e,i}$ in Eq.(7) indicates the distance between the origin $(0,0)$ and experimental value $(Q_{e,i},\ H_{e,i})$ in Figure 5. In addition, $\delta_{t,i}$ indicates the distance between the origin $(0,0)$ and the point $(Q_{t,i},\ H_{t,i})$ that is the intersection between the line connecting the origin $(0,0)$ and experimental value $(Q_{e,i},\ H_{e,i})$, and the theoretical flow rate characteristic line. In addition, n is the number of samples used for calculating the average relative error δ_{ave}, and it was set to the same value as the number of plots in experiment. It was also supposed that theoretical flow rate formula Eq.(6) was applicable when $E_{ave} \leqq 30\%$.

4.1. Experiment varying the oil groove length L

Figure 6 shows the relationship between the oil groove length L and average relative error δ_{ave} derived from the results of the experiment which evaluated the flow characteristics by varying the oil groove length L and calculated each average relative error δ_{ave}. Based on Figure 6, it was found that the difference between the experimental values and theoretical values became larger at low rotational speed when the oil groove length L was shorter, although they showed a relatively good match at high rotational speeds. These phenomena indicate that Assumption (1), set up when the theory was discussed, holds for a shorter oil groove length L at higher rotational speeds than lower ones. It was also found based on Fig. 6 that application of the theoretical formula Eq.(6) was possible with $E_{ave} \leqq 30\%$ when the oil groove length $L \geqq 60\text{mm}$.

4.2. Experiment varying oil groove angle β

Figure 7 shows the relationship between oil groove angle β and average relative error δ_{ave} derived from the results of the experiment which evaluated the flow characteristics by varying the oil groove angle β. Based on Figure 7, it was found that the experimental value and theoretical value was approximately matched in the high rotational speed from low rotational speed. While Figure 7 also describes oil groove length L, it was found that $E_{ave} \leqq 30\%$ was obtained for oil groove length $L \geqq 60\text{mm}$, making it possible for a theoretical flow rate formula Eq.(6) to be applied. Based on these, it was found that whether the theoretical flow rate formula Eq.(6) is applicable or not is dependent on the oil groove length L, not on the oil groove angle β.

4.3. Experiment varying the oil groove cross section shape

Figure 8 shows the relationship between oil groove aspect ratio h/b and average relative error δ_{ave} derived from the results of the experiment which evaluated the flow characteristics by varying oil groove width b and depth h with the oil groove cross section kept constant. Based on Figure 8, it was found that the difference between the experimental value and theoretical value became larger when the oil groove aspect ratio h/b was larger at any rotational speed N. It is assumed that this was because the oil flow came under a larger influence of the side wall when the oil groove width b was smaller if aspect ratio h/b was larger, making it impossible to apply Assumption (2), in which the oil flow is uniform *x*-*y* flow in z direction. It was also found that application of the theoretical formula was possible at $E_{ave} \leqq 30\%$ when aspect ratio was $h/b \leqq 0.5$.

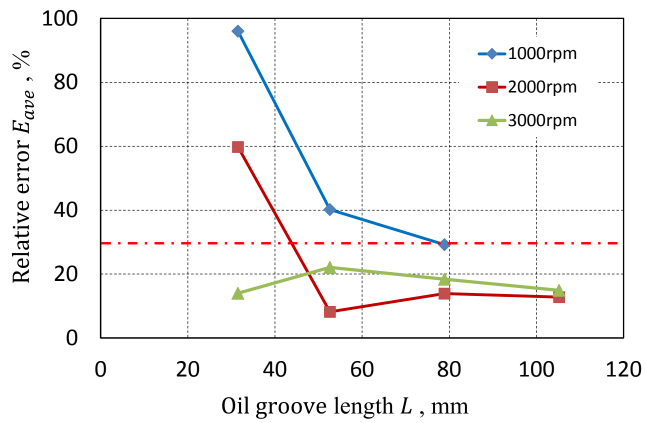

Figure 6: Relationship between oil groove length and average relative error

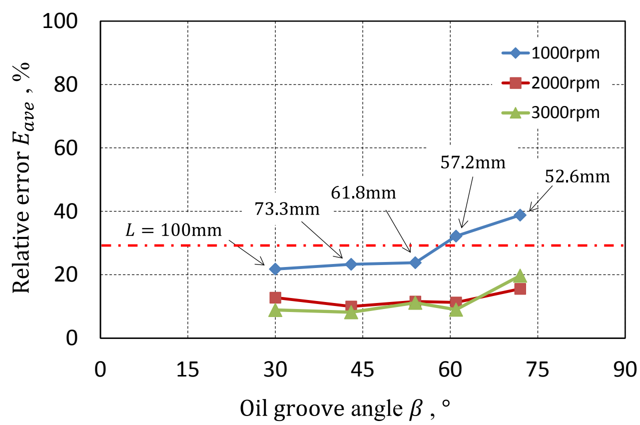

Figure 7: Relationship between oil groove angle and average relative error

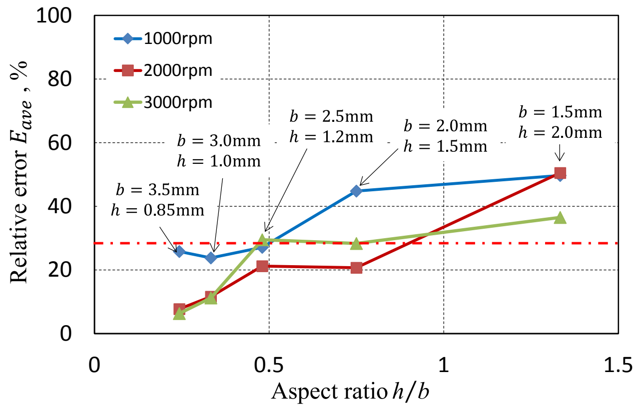

Figure 8: Relationship between oil groove aspect ratio and average relative error

5. CONCLUSION

We conducted a theoretical study of fluid dynamics on the flow characteristics of the oil viscosity pump with a spiral groove on the surface of the shaft opposite to the bearing and verification experiments using a model test machine, and found the followings:

(1) When the oil flow in a spiral groove is assumed to be a "uniform, two-dimensional flow which is a steady and sufficiently developed laminar flow," the theoretical flow rate Q_t in the oil viscosity pump and the pump head H can be expressed with a linear function formula. This relationship was also verified in experiments.

(2) Whether the theoretical flow rate formula can be applied does not depend on the oil groove angle β, but on the oil groove length L. It is possible to estimate the flow characteristics with average relative error $E_{ave} \leqq 30\%$ if the oil groove length $L \geqq 60\text{mm}$.

(3) Whether the theoretical flow rate formula can be applied depends on the oil groove depth h if the oil groove has a rectangular cross section. It is possible to estimate the flow characteristics with average relative error $E_{ave} \leqq 30\%$ if the oil groove aspect ratio $h/b \leqq 0.5$.

NOMENCLUTURE

b	Oil groove width	(m)
E_{ave}	Average relative error	(%)
F_c	Centrifugal force	(N)
g	Gravity	(m/s^2)
h	Oil groove depth	(m)
H	Oil pump head	(m)
L	Oil groove length	(m)
L_s	Shaft length	(m)
n	Sample number	(−)
N	Shaft rotating speed	(rpm)
Q	Oil flow rate	(m^3/s)
r_s	Shaft radius	(m)
u	Oil velocity of x direction	(m/s)
U	Relative velocity of bearing wall	(m/s)
v	Oil velocity of y direction	(m/s)
β	Oil groove angle	(°)
δ	Distance in Figure 5	(mm)
ν	Oil kinematic viscosity	(m^2/s)
ρ	Oil density	(kg/m^3)
Subscript		
e	Experimental	
t	Theoretical	

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